Automotive internal combustion engine with bearings structure for crankshaft and outputshaft.

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Erfinder :	OKUI KAORU (JP); KOBAYASHI MANABU (JP)
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Applicant: YAMAHA HATSUDOKI KABUSHIKI KAISHA
2500 Shingai
Iwata-shi Shizuoka-ken, 438(JP)

Inventor: Okui, Kaoru
161-158 Toshin-cho Iwata-shi
Shizuoka-ken, 438(JP)
Inventor: Kobayashi, Manabu
4-1-3 Fujimi-cho Iwata-shi
Shizuoka-ken, 438(JP)

Representative: Patentanwälte Grünecker, Kinkeldey, Stockmair & Partnerner
Maximilianstrasse 58
D-8000 München 22(DE)

Automotive internal combustion engine with bearings structure for crankshaft and outputshaft.

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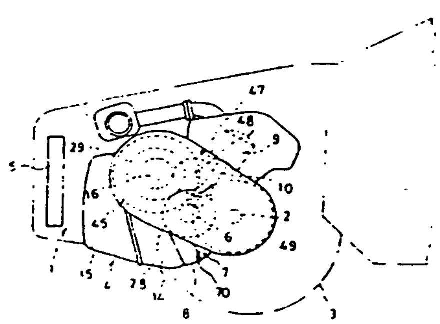

FIG. 1

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INTERNAL COMBUSTION ENGINE FOR AUTOMOTIVE VEHICLES

The present invention relates to an internal combustion engine for automotive vehicles comprising a crankshaft having an output gear and a power output shaft disposed in parallel to the crankshaft adapted to distribute the output power of the crankshaft to clutch and or transmission means and/or accessory equipment.

Specifically, the present invention refers to a multi-cylinder engine having a compacted structure due to the provision of an output power shaft rotatably supported on the same side and above the crankshaft which is in engagement with said output shaft in order to transmit output power to a flywheel associated to a manually operable transmission means or a torque converter of an automatic transmission, associated to one end of the output shaft.

Nowadays, automotive vehicles are provided either with a manually operated transmission or an automatic transmission. In case a manually operated transmission is selected, the internal combustion engine comprises a fly wheel which, instead of being supported by the crankshaft itself, could be disposed on one end of an output shaft which is disposed slightly above the crankshaft on the same side of the engine block and is driven by the crankshaft. In this way, it becomes possible to shorten the extent of the engine in the direction of the crankshaft axis. Of course, the same applies with respect to the location of a torque converter of an automatic transmission disposed on one end of the output shaft. Again, the output shaft is rotatably supported slightly above the crankshaft and, moreover, serves to drive a plurality of other systems and auxiliary equipment such as alternator, oil and water pump means and the intake and exhaust valve operating cam mechanism. Hereupon, it is usual to provide an appropriate backlash in between an output gear of the crankshaft which is in mesh with a counter-gear on the output shaft in order to transmit the output power from the crankshaft to the output shaft. Said backlash in between said two gears increases the reliability of the system so that seizures may be prevented.

However, if a considerable stress is applied on the bearing portions of the crankshaft supporting structure, including the main bearing cap sections, or is applied to the output shaft and its associated supporting structure, some positional offset may be caused in between both shafts resulting in an undesirable large backlash in between the drive gears.

In particular, if a flywheel or a torque converter is mounted on the output shaft, there is a fear that, when the backlash in between the transmission gears of both shafts has excessively increased, undesirable extra gear noise is generated considering that the rotation of the crankshaft is transmitted from the output gear of the crankshaft to the counter-gear of the output shaft without being relieved of any speed fluctuations.

Accordingly, an improved engine structure should be developed enabling to gain the merits of a reduced length of the engine block in the axial direction of the crankshaft while preventing the transmission in between the crankshaft and the output shaft from generating undesirable gear noise.

Moreover, when redesigning the lower portion of the engine block, intending to provide an improved rigidity of the bearing structure adapted to support the output power transmitting elements including redesigning the power transmitting elements in between the crankshaft and the output shaft, care should be taken to maintain the balance for each cylinder with respect to the up-down movement of the piston and the rotation of the crankshaft.

Conventionally, on the crankshaft for a straight-type multi-cylinder engine, a pair of crankarms supporting the associated crankpin for each cylinder (which, in turn, supports the connecting rod, respectively) is designed as a counterweight so that the reciprocational and rotational balances for each cylinder may be obtained. Thus, normally, all counterweights can be of identical shape.

If, on the other hand and in compliance with the above design considerations, an output gear is to be provided on the crankshaft in order to shorten its total length, only one of the crankarms offers to be shaped as a gear in mesh with a counter-gear on the output shaft which leads to the requirement that, in this case, the balances for the related cylinder must be obtained only through that other crankarm holding the crankpin at the side opposite to the crankarm which has become a gear. Thus, in such cases, difficulties arise to reliably obtain the required balances between all cylinders as the balances for the other cylinders are obtained through the normal two counterweights, whereas the balances for the cylinder having a geared crankarm must be obtained through only the other one counterweight. Moreover, it has also to be borne in mind that the specific design of one counterweight associated to the output gear with a crankpin in between is undesirable under the viewpoint of manufacturing convenience as the entirety of counterweights can no longer have an identical form but one crankarm mating with the gear must be formed differently from those of the other cylinders.

Accordingly, it is an objective of the present invention to improve the rigidity of a power transmitting structure of an internal combustion engine as indicated above, in order to maintain a predetermined given distance in between a crankshaft axis and an associated output shaft under all operating conditions of the engine, thus, reducing transmission noise radiated from the power transmission in between the crankshaft and an output shaft which lends assistance in rendering the engine more compact.

In order to achieve the afore-mentioned objective, according to the present invention, an internal combustion engine as indicated in the introductory portion of the description is characterized in that the crankshaft and the output shaft both are supported by a common integral unitary bearing structure.

According to an advantageous embodiment of the present invention, the bearing structure for journaling the crankshaft and the power output shaft is composed of an upper half which preferably is integral with the crankcase of the engine providing a unitary cylinder block structure, and a lower half which is bolted to the upper half and associated cylinder block structure wherein the upper and lower halves each form an integral bearing structure for rotatably supporting the crankshaft and the output shaft.

According to yet another advantageous embodiment of the present invention, the crankshaft, supported by the upper and lower halves of the unitary bearing structure, has a pair of crankarms for each cylinder wherein each pair of crankarms is composed of a disk and a counterweight with a crankpin in between. Moreover, one of the disk-shaped crankarms is designed to form a gear which forms an output gear of the camshaft which is in mesh with a counter-gear of the output shaft.

Further advantageous embodiments of the present invention are laid down in the other subclaims.

Since the present invention provides for the crankshaft and power output shaft to be supported integrally by an upper and lower half of the bearing structure, the rigidity of the bearing portion rotatably supporting the crankshaft and the power output shaft can be improved. Accordingly, even if a big stress is applied on the bearing portion of the crankshaft and/or the output shaft, a positional offset in between both shafts can be minimized to maintain an appropriate backlash in between the output gear of the crankshaft and the counter-gear of the output shaft, thus, preventing seizure of the gears and undesirable gear noise generation to occur.

Moreover, the advantageous embodiment of the present invention concerning the design of the crankarm structure of the crankshaft enables to increase the thickness of the output gear of the crankshaft in order to provide a smoothly operating gear transmission in between the crankshaft and the output shaft. Finally, even though each of the pairs of crankarms per cylinder comprises a disk and a counterweight, all the counterweight can be of identical shape facilitating the manufacturing of the crankshaft and, moreover, the balances of the cylinders can be assured even though one of the crankarms is designed to form an output gear which is in mesh with a counter-gear of the crankshaft.

Further objectives, features and advantages of the present invention will become more apparent from the following description of specific embodiments of the present invention in conjunction with the associated drawings, wherein:

Figure 1 is a side view showing the assembled condition of an internal combustion engine according to an embodiment of the present invention.

Figure 2 is a plan view of Figure 1.

Figure 3 is a side view of the internal combustion engine as in Figure 1, but in a larger scale and with some connected parts (e.g. the transmission) removed.

Figure 4 is a front view of the internal combustion engine as shown in Figure 1.

Figure 5 is a front view of an oil pan wherein the oil cooler, an oil filter and the oil tank have been omitted.

Figure 6 is a sectional view of the engine along the line VI-VI in Figure 5.

Figure 7 is a sectional view along the line VII-VII in Figure 6.

Figure 8 is a sectional view along the line VIII-VIII in Figure 6.

Figure 9 is a perspective view of the lower half of the bearing structure of the engne with the mounting holes omitted.

Figure 10 is a back view of the oil pan according to Figure 5 viewed from its backside.

Figure 11 is a partial sectional view similar to the right-hand side view of Figure 7 but with an automatic transmission associated to an output shaft of the engine.

Figure 12 is a plan view of the crankshaft as shown in Figure 7, and

Figures 13 to 15 are sectional views along the lines II-II, III-III and IV-IV of Figure 12, respectively.

In Figures 1 and 2, the reference number 1 denotes the engine compartment of an automobile formed above and between the right and left front wheels 3 connected through front wheel shafts 2. Within this engine compartment 1 is mounted an engine unit 4 having a 4-stroke 6-cylinder internal

combustion engine with its radiator 5 arranged in front of this engine unit 4. The passenger compartment has been made spacious by disposing the engine crankshaft 6 laterally of the vehicle.

The cylinder block 7 is composed of a crankcase 7a and a cylinder body 7b integrally formed with cast iron. The pistons 9 provided in the cylinder body 7b are each connected with the crankshaft 6 through respective connecting rod 10. On the cylinder block 7 is mounted a cylinder head 11 provided with a head cover 12. and an ignition plug is provided on each cylinder.

The cylinder bank of the engine is inclined backwardly relative to the vertical of the vehicle. The power output shaft 16 for distributing the output power of the crankshaft 6 is disposed in parallel with the crankshaft 6 and is disposed slantly forwardly above and close to the crankshaft 6. The oil tank 15 reserving engine oil is located slantly forwardly under the crankshaft 6 and the output shaft 16 and is thus faced forwardly of the vehicle as shown by the arrow mark FWD in Figure 3 The ouput shaft 16 is positioned in such a manner that the angle α formed between the cylinder axis plane L1 and the plane L2 including both of the crankshaft 6 axis and the power takeout shaft 16 axis may be an acute angle.

The structure of the crankshaft 6 is explained in greater detail below with respect to Figures 12 to 15. Here. it should only be noted that the crankshaft 6 has an output gear 28 formed around one of its crankarms. and this gear 28 is in engagement with a counter-gear 29 mounted on the power takeout shaft 16. Further. a gear 30 provided on the power takeout shaft 16 is connected with a gear 32 on the countershaft 31 journaled on the cylinder head 11 through a first chain 33. The gear 34 mounted on the countershaft 31 is connected to the gear 38 on the camshaft 37 is rotated by the rotation of the crankshaft 6. and the cams 39 are rotated together with the camshaft 37 and operate the intake and exhaust valves (not shown) with predetermined timings.

As shown in Figures 1 and 7. on one end of the ouput shaft 16 is provided a flywheel 45 and a clutch mechanism (not shown) so that the power may be transmitted to the front wheel shafts 2 for front wheels 3 through a transmission 47 constituting a manual transmission. The flywheel 45 is positioned at one end of the crankcase 7a longitudinally of the crankshaft in such a manner that it will not project outwardly of this crankcase end. As shown in Figure 1. the primary side of the transmission 47 is disposed on the power takeout shaft 16. and the secondary side is disposed on a countershaft 48 to rotate the front wheel shaft 2 through a gear 49 provided on the wheel shaft 2.

On the other end of the output shaft 16 is provided an auxiliary drive pulley 50 as shown in figure 7. which pulley 50 is accomodated within a concave 51 formed laterally on the crank bearing 60 of the crankcase 7a so that it may not project outwardly of this crankcase end. A belt 55 is wrapped around this auxiliary drive pulley 50 and the auxiliary pulleys for auxiliaries such as alternator 52. power steering pump 53. air compressor 54. etc.. so that these auxiliaries are simultaneously driven by the rotation of the power output shaft 16.

The bearing portion for journaling the crankshaft 6 and the power takeout shaft 16 is composed of an upper half 70 formed integrally with the crankcase 7a and a lower half 8. and the upper half 70 and the lower half 8 integrally journal each of the crankshaft 6 and the power takeout shaft 16.

The upper half 70 has bearings 60 for the crankshaft 6 formed for each cylinder. bearings 61 for the power takeout shaft 16. and further openings 62 for the oil passage 223 formed on its bottom side. The upper half 70 may be formed as a separate member although it is formed integrally with the crankcase 7a in this embodiment

As shown in Figure 8 and 9. the lower half 8 is made of cast iron. and has bearings 80 for the crankshaft 6. bearings 81 for the power takeout shaft 16. and further oil passage openings 8a formed on its bottom side. As shown in Figure 7. this lower half 8 is fastened singly on the cylinder block 7 on which is integrally formed the upper half 70 through mounting bolts 400 at twenty-two points. and the oil pan 14 is fastened on the upper half 70 together with the lower half 8 through mounting bolts 401 at fifteen points. and further is fastened singly on the lower half 8 through mounting bolts 402 at five points.

As is shown in Figures 7 and 8. the axial length of a bearing section (X:X′) of the upper and lower halves 70:8 respectively adapted to rotatably supporting the output shaft 16 is designed to be shorter than the length of a bearing section (Y:Y′) of the upper and lower halves 70:8 respectively adapted to rotatably support the crankshaft 6. Thus the output shaft bearing structure is more lightweight than the crankshaft bearing structure. The afore-indicated design considerations render it possible to make the journal caps compact and lightweight. Moreover. the assembly of the bearing structure is facilitated as the heavier crankshaft supporting portion Y:Y′ is arranged lower than the lighter output shaft supporting portion. Thus. it is easy to affix the journal caps to the engine.

Moreover. as is apparent from a comparison of Figures 7 and 8. the bearing section X for rotatably supporting the output shaft 16 which is designed to be integral with the upper half 70 is shorter than the cooperating bearing section X′ for rotatably supporting the output shaft 16 which is integrally

provided at the lower half 8.

Since the lower half 8 is a member separate from the oil pan 14, the crankshaft 6 and the power takeout shaft 16 can be left journaled by the lower half 8 even when the oil pan is removed for maintenance operations such as renewal of the strainer 25, which serves to facilitate engine maintenance operations.

Since the crankshaft 6 and the power takeout shaft 16 are integrally journaled by the bearing portion composed of the upper half 70 and the lower half 8 as described above, the rigidity of the bearing portion is increased. Therefore, positional offset is restrained from occurring between the crankshaft 6 and the power takeout shaft 16 even when being subjected to greater stress transmitted to the bearing portion of the shafts 6,16 and the fluctuation of the backlash provided between the gears 28 and 29 on the crankshaft 6 and the output shaft 16, respectively, can be restrained.

Accordingly, since the backlash between gears 28 and 29 can be kept appropriate even when the flywheel is provided on the output shaft 16 and, therefore, the rotational speed fluctuation of the crankshaft 6 is transmitted to the output shaft 16 without being relieved. Then, noise occurrence between gears 28 and 29 can be reduced.

Further, when the lower half 8 is formed of cast iron (although it may be formed of aluminium casting) as in this embodiment, the distance of the bearing 80 for the crankshaft 6 and the bearing 81 for the power takeout shaft 16 can also be restrained from getting increased by the deformation of the lower half 8 caused by the effect of the engine heat, and thus the backlash between gears 28 and 29 on the crankshaft 6 and the power takeout shaft 16, respectively, can be kept appropriate.

In the bottom of the crank chamber A formed with the upper half 70 on the cylinder block 7, the lower half 8 and the oil pan 14 is collected the engine oil after lubricating various portions of the engine such as the cylinder block 7, etc.

The oil pan 14 and the oil tank 15 are both formed of aluminium casting, and are partitioned from each other by a wall portion 14a, and the oil pan 15 is mounted on the tank mounting seat 14k of the oil pan 14.

P is a pump unit having a discharge pump for sending the oil collected in the crank chamber A to the oil tank 15 and a oil feeding pump for supplying the oil collected in the oil tank 15 to various portions of the engine combined integrally with each other. This pump unit P is mounted on the pump mounting portion B formed on the oil pan 14 and on the pump mounting portion C formed on the upper half 70 and the lower half 8. The oil passages for communicating the discharge pump and the oil feeding pump with the oil pan 14 and the oil tank 15 are formed integrally with the oil pan 14.

Figure 11 is a partial sectional view of a vehicle engine unit showing another embodiment.

In that case, the engine unit is provided with an automatic transmission. As no further modifications have been made, except for said exchange of the transmission unit, the description of the engine is not repeated.

On one end of the output shaft 16 is secured a drive plate 500 through bolts 501, and on this drive plate 500 is fixed, through bolts 504, a pump impeller 503 for the torque converter 502. Opposite to this pump impeller 503 is disposed a turbine runner 505, which is spline-engaged with a transmission input shaft 506, and a stator 507 is provided on the stator shaft 509 through a one-way clutch 508.

Accordingly, the rotation of the crankshaft 6 is transmitted to the output shaft 16 through gears 28 and 29, and then to the torque convertor 502 through the drive plate 500. Here, the turning effort of the pump impeller 503 is transmitted to the turbine runner 505 through fluid, through which the transmission input shaft 506 is rotated and transmits the rotation to the front wheel shaft through a planetary gear unit (not shown).

This torque convertor 502 constitutes an automatic transmission together with a planetary gear unit, and relieves the crankshaft rotation of its speed fluctuation like a flywheel 45 in the above-described first embodiment.

Turning now to the crankshaft structure, as shown in Figures 12 to 15, it is to be noted that the crankshaft 6 is supported to journal bearings on both sides of each crankpin 12A to 12F corresponding to each cylinder. The reference numerals 14A to 14G in Figure 12 denote journal portions rotatably supported by these bearings. The crankpins 12A to 12F are connected to the journal portions 14A to 14G through crankarms 16A to 16L for each cylinder wherein each pair of crankarms associated to a cylinder comprises a circular disk 16D,16E,16H and 16L, said disks having a relatively small thickness while the other respective crankarm of each of the pair of crankarms is designed to form a conventional counterweight 16B,16C,16F,16G,16J, respectively. Thus, the crankarm pairs 16B and 16C, 16F and 16G, 16J and 16K are positioned on both sides of the journal portions 14B,14D and 14F respectively.

One of the disk-shaped crankarms 16E is shaped to form the gear 28 which is in mesh with the counter-gear 29 shown in phantom lines in Figure 12.

In order to reduce the gear noise, enabling smooth operation of the output gear 28 and, more-

over, to increase the strength of the output drive gear 28, the thickness of said gear 28 has been increased. In order to compensate unbalances for the associated cylinder and the plurality of cylinders which would tend to arise from the output gear 28 having an increased thickness, the associated disk 16H at the opposite side of the journal portion 14E, in its thickness, is considerably reduced to form a drive plate, thus enabling to increase the widths of the drive gear 28 without losing the crankshaft balance.

According to the above-identified layout of the crankshaft and associated crankarms, it is apparent that to each cylinder corresponds one disk 16A,16D,16E,16H or 16L while the one "disk" 16I is shaped to form a relatively thick output gear 28 and, moreover, to each cylinder corresponds one counterweight 16B,16C,16F,16G and 16J. In this way, since the disks 16A,16D,16E,16H,16L and the gear 28 ("disk 16I") all have even circumferential weight distributions, the balance for the reciprocating parts of each cylinder (piston, counterweight 16B,16C,16F,16G and 16J. Moreover, since the balance for every cylinder is obtained through one disk or, in one case through a gear, and one counterweight, balances for all cylinders can be obtained well under the same conditions for each of the cylinders and an appropriate crankshaft balance is obtained. Moreover, all counterweights 16B,16C,16F,16G and 16J are of identical form which is convenient for the design and manufacturing of same. I.e., since these counterweights 16B,16C,16F,16G,16J are of identical form, balances of the entirety of cylinders will be obtained under the same condition and, accordingly, it is sufficient to design one counterweight for any of the cylinders in that way, that balance for said cylinder may be well obtained and, on the basis of said sample design, it is only required to manufacture the further counterweights having an identical shape.

The circular shape of the disks fascilitate the machining of same. In this case, not only that disk 16H opposite to the gear 28 at the crankshaft journal portion 14A is formed in the shape of a plate deviating from the other disks 16D or 16E but also the disk-shaped crankarms 16A and 16L of the outer pairs of crankarms form similar plate members. In this way, the length of the crankshaft 6 and the associated crankcase structure can be reduced without effecting the crankshaft balance. It is not necessary that each of the outer pairs of crankarms comprises such a plate disk member 16A, 16L but at least one of said outer pairs of crankarms is designed in this way.

In the embodiment of the present invention as shown, the disks 16A and 16L are of identical plate form while the disks 16D and 16E are of a different but identical form among each other. The counterweights 16B and 16K, 16C and 16J, 16F and 16G are at the same phase, respectively.

Moreover, since the disks 16A,16D,16E,16H and 16L are made to have only a small thickness, it is possible to increase the width of the gear 28 ("disk 16I") while keeping the distance in between the cylinders of the cylinder bank constant.

Accordingly, the thickness of the at least one outer plate-like disk 16A,16L can be reduced without losing the balance. By reducing the thickness of the outer disk 16A, 16L the length of the engine in axial direction of the crankshaft axis can be reduced even though the location of the centerline of the associated outer crankpin cannot be moved, i.e., the length l can be reduced even though a length s is fixed and cannot be varied.

Specifically, as shown in Figure 12, the sum of the distances B and C between the crankpins 12D and 12E and the journal portion 14E (B+C) is twice the sum of the distances a + a between another crankpin and its adjacent journal portion (2a = b + c). In this way, the widths of the teeth of the gear 28 can be increased by reducing the thickness of the disk 16H resulting in B smaller than A smaller than C. Thus, the enlarged width of the gear 28 provides a reduced gear noise and capability for transmitting bigger torque from the gear 28 to the gear 29. The distance G is greater than the distance H even though the distance in between the crankpins are equal and fixed.

As a result of the afore-indicated design considerations, the hammering noise of the teeth of the gear 28 ("disk" 16I) can be reduced and the strength of the gear can be increased. Since thinner plate-like disks 16A,16L are positioned at both ends of the crankshaft, the crankcase can be made more compact. Since, on both sides of a journal portion, in this embodiment, fellow counterweights or fellow disks are disposed, the balance of the couple forces of each journal bearing can also be improved.

As shown in dotted line in Figure 12 and shown in full lines in Figure 13 to 15 for a main end bearing section of the crankshaft and two intermediate bearing sections, oil passages 18,18A,18B for press-feeding the lubricating oil to the journal portion 14 and to the crankpin 12 are shown. Although these oil passages are formed for each of the cylinders, they are only shown on the left-hand side in Figure 12.

Although, in this embodiment, journal portions 14 on both sides of each cylinder are provided through the crankshaft, some journal portions could also be omitted if the load operating conditions of the crankshaft allow such a design to be selected.

By means of the present invention, a couple of merits can be gained as follows:

Since the internal combustion engine comprises a crankshaft having an output gear formed on it and a power output shaft having a countergear intermeshing with the output gear of the crankshaft and a flywheel or a torque converter for a manual or automatic transmission is provided on this output shaft, the crankshaft needs not to be adapted to support said fly wheel or said torque converter resulting in a considerable reduced length of the engine in a direction parallel to the crankshaft axis.

Moreover, since the bearing portion for journaling the crankshaft and the power output shaft is composed of an upper half and a lower half, each of them forming an integral unitary bearing structure for each commonly supporting the crankshaft and the output shaft, the rigidity of the bearing portion is improved and a positional offset in between the crankshaft and the power output shaft can be reduced even if considerable stress is applied on the bearing portions of both shafts. Accordingly, the backlash of the intermeshing gears on the crankshaft and the power output shaft can be kept appropriate even when the flywheel or a torque converter is supported by the output shaft. Thus, even though rotational speed fluctuations of the crankshaft are transmitted to the output shaft without being relieved, noise occurrence in between both gears (gear transmission noise) can be reduced.

Noise generation can also be reduced by increasing the widths of the crankshaft gear enabling to increase the amount of torque transmitted from the crankshaft to the output shaft. Said objective is obtained by an advantageous specific design of the crankarms of the crankshaft in as far as one crankarm of each pair of crankarms holding an associated crankpin is formed of a circular disk having an even circumferential weight distribution while the other crankarm of every pair of crankarms is formed to be a counterweight having an uneven circumferential weight distribution. The output gear of the crankshaft is formed as a substitute of one of the circular disk crankarms and the balance for each cylinder is obtained through the one counterweight for each cylinder.

Finally, all counterweights can be of identical form and it is not necessary to design differently said counterweights for each cylinder resulting in reduced manufacturing costs. By means of designing the disk-shaped crankarms of the outer pairs of crankarms to form plates, the overall length of the crankcase can be reduced rendering the engne more compact.

Claims

1. An internal combustion engine for automotive vehicles comprising a crankshaft having an output gear and a power output shaft disposed in parallel to the crankshaft adapted to distribute the output power of the crankshaft to clutch and/or transmission means and/or to accessory equipment,
characterized in that
said crankshaft (6) and said output shaft (16) both are supported by a common integral unitary bearing structure (70,8).

2. An internal combustion engine as claimed in Claim 1,
characterized in that
that said bearing structure for journaling the crankshaft (6) and the power output shaft (16) is composed of an upper half (70) and a lower half (8), each forming an integral crankshaft output shaft journaling structure.

3. An internal combustion engine as claimed in Claim 2,
characterized in that
the upper half (70) is integral with a lower crankcase portion of the cylinder block (7) whereas the lower half (8) is bolted to the integral upper half (70) of the cylinder block (7).

4. An internal combustion engine as claimed in Claim 3,
characterized in that
the lower half (8) forms a member separate from an oil pan (14) which is fastened together with the lower half (8) to the upper half (70) which is integral with the crankcase (7a) as formed by the lower portion of the cylinder block (7).

5. An internal combustion engine as claimed in Claim 2,
characterized in that
the upper half (70) provides bearings (60) for the crankshaft (6) formed for each cylinder, bearings (61) for the output shaft (16) and further openings (62) for an oil passage (223) formed on the bottom side of the upper half (70).

6. An internal combustion engine as claimed in Claim 2 or 5,
characterized in that
the upper half (70) is a member separate from the cylinder block (7).

7. An internal combustion engine as claimed in at least one of the preceding Claims 1 to 6,
characterized in that
a cylinder block (7) of the engine is composed of a crankcase (7a) and a cylinder body (7b) which are integral to each other.

8. An internal combustion engine as claimed in at least one of the preceding Claims 1 to 7,
characterized in that
the lower half (8) is mado of cast-iron or aluminium alloy and provides bearings (80) for the crankshaft (6) bearings (81) for the output shaft (16) and for

the oil passage openings (8) formed on the bottom side of the lower half (8).

9. An internal combustion engine as claimed in at least one of the preceding Claims 1 to 8, **characterized in that** at its one end, the output shaft (16) supports an accessory drive pulley (50) disposed opposite to a crankshaft bearing (60,61) and, at its other end, supports a clutch and/or transmission means (45,500).

10. An internal combustion engine as claimed in at least one of the preceding Claims 1 to 9, **characterized in that** the output shaft (16) is disposed slightly forwardly above the crankshaft (6) in such a manner that a plane (L2) including both the axis of the crankshaft (6) and the axis of the output shaft (16) intersects with a plane (L1) including the cylinder axis of the cylinder bank at an acute angle ( ).

11. An internal combustion engine as claimed in at least one of the preceding Claims 1-10, **characterized in that** a bearing section of the upper and lower half (70,8) rotatably supporting the output shaft is designed to have a length shorter than the length of the bearing section adapted to rotatably support the crankshaft (6).

12. An internal combustion engine as claimed in any of the preceding Claims 1-11, **characterized in that** the bearing section for rotatably supporting the output shaft (16) designed to be integral with the lower half (8) is shorter than the co-operating bearing section for rotatably supporting the output shaft (16) as integrally provided at the upper half (70) which, preferably, is integral with the cylinder block (7).

13. An internal combustion engine as claimed in Claims 11 or 12, **characterized in that** the bearing section rotatably supporting the output shaft (16) has a weight lower than that of the bearing section adapated to rotatably support the crankshaft (6).

14. An internal combustion engine as claimed in at least one of the preceding Claims 1-13, **characterized in that** the crankshaft (6) provides a pair of crankarms (16A to 16L) for each cylinder, each pair of crankarms holding each of the crankpins comprising a disc (16A, 16D, 16E, 16H and 16L) and a counterweight (16B, 16C, 16F, 16G and 16J).

15. An internal combustion engine as claimed in Claim 14, **characterized in that** one of the disc-shaped crankarms is shaped to form a gear (28) which is in mesh with a counter-gear (29) mounted on the output shaft (16).

16. An internal combustion engine as claimed in Claim 15, **characterized in that** each disc-shaped crankarm (16A, 16D, 16E, 16H and 16L) is formed to be a circular disc having an even circumferential weight distribution while the associated counter-weight (16B, 16C, 16F, 16G and 16J) of each of the pairs of crankarms have an uneven circumferential weight distribution.

17. An internal combustion engine as Claim 16, **characterized in that** the output gear (28) of the crankshaft which is in mesh with the counter-gear (29) of the output shaft (16) has a width which is thicker than that of the circular discs (16A, 16D, 16E, 16H and 16L) comprised in the other pairs of crankarms and, preferably, exceeds the thickness of the counter-weights (16B, 16C, 16F, 16G and 16J) of the pairs of crankarms.

18. An internal combustion engine as claimed in at least one of the preceding Claims 14-17, **characterized in that** an opposite disc (16H) associated to the same journal portion (14A) as the gear (28) has a width which is thinner than those of the other discs (16D, 16E and 16H) comprised in pairs of crankarms disposed in between the outer pairs of crankarms (16A, 16B; 16K, 16L) on both ends of the crankshaft (6).

19. An internal combustion engine as claimed at least one of the preceding claims 14-18, **characterized in that** the discs (16A, 16L) of the pairs of outer crankarms (16A, 16B; 16K, 16L) have an axial width which is thinner than those of the other disc-shaped crankarms (16D, 16E) with the exception of the one disc (16H) which is disposed opposite to the gear (28) at the same journal bearing (14E).

20. An internal combustion engine as claimed in at least one of the preceding Claims 14 to 19, **characterized in that** the distance (c) in between the center of the crankpin (12E) held by the gear (28) and a journal portion (14E) associated to the gear (28) is greater than the distance (b) between said journal portion (14E) to a next crankpin (12D) held by the disk (16H) disposed opposite to the gear (28) even though the distances (2a) between the centers of the crankpins (12A to 12F) are equal and constant.

21. An automotive internal combustion engine as claimed in Claim 20, **characterized in that** the distance in between adjacent crankshaft bearing portions (14A,14F) disposed at both sides of said pair of crank arms (16I,16J), which comprise the output gear (28) of the crankshaft (6) is greater than the distance in between other neighboured journal portions (14A,14B,14C,14D) supporting the

crankshaft (6).

22. An internal combustion engine as claimed in at least one of the preceding Claims 1 to 21.
characterized in that
each journal portion (14A to 14G) comprises an oil passage (18.18A.18B) communicated by inclined channels to the adjacent crank pin (12A to 12F) press-feeding lubricating oil to the journal portions (14A to 14G) and to the crank pins (12A.12F) respectively.

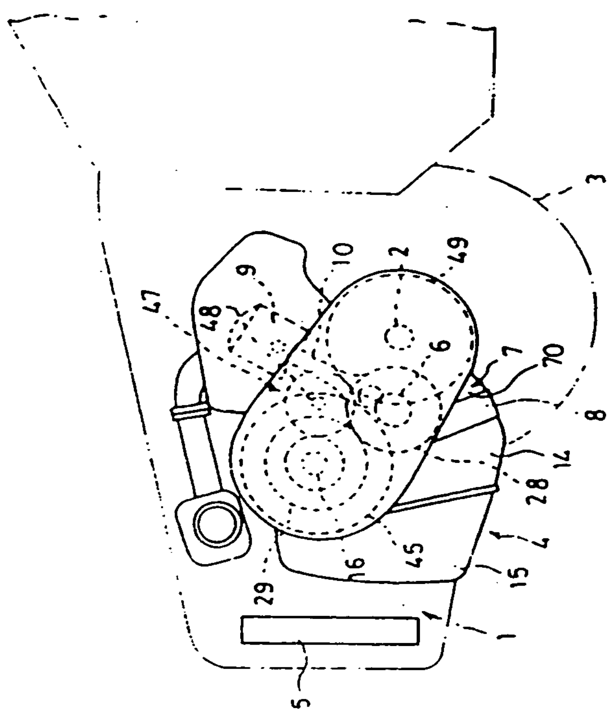


FIG. 1

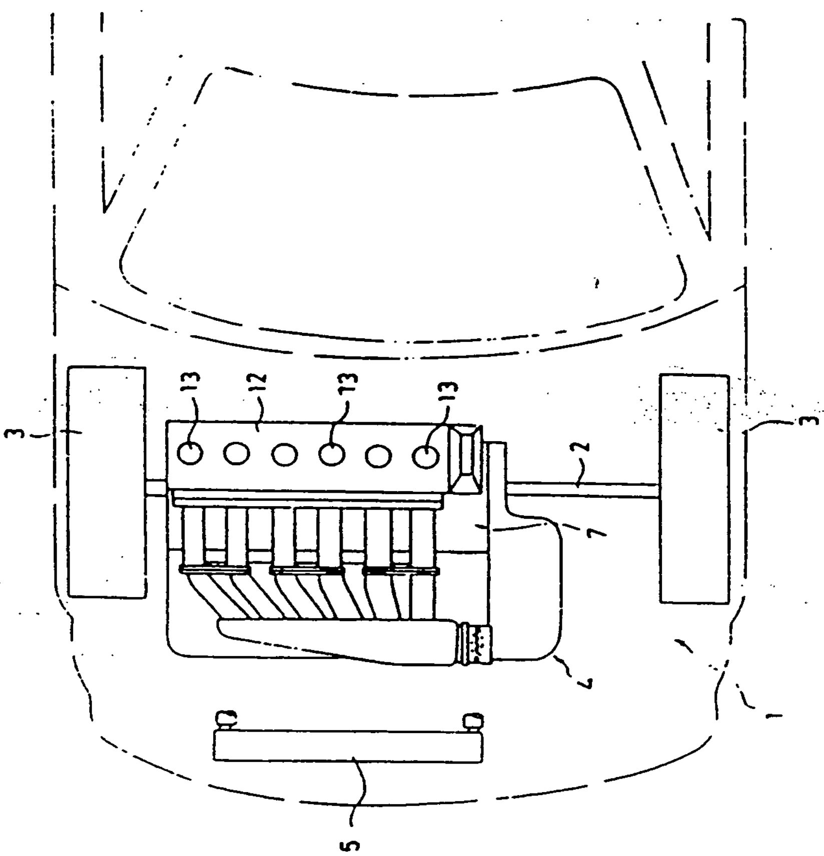
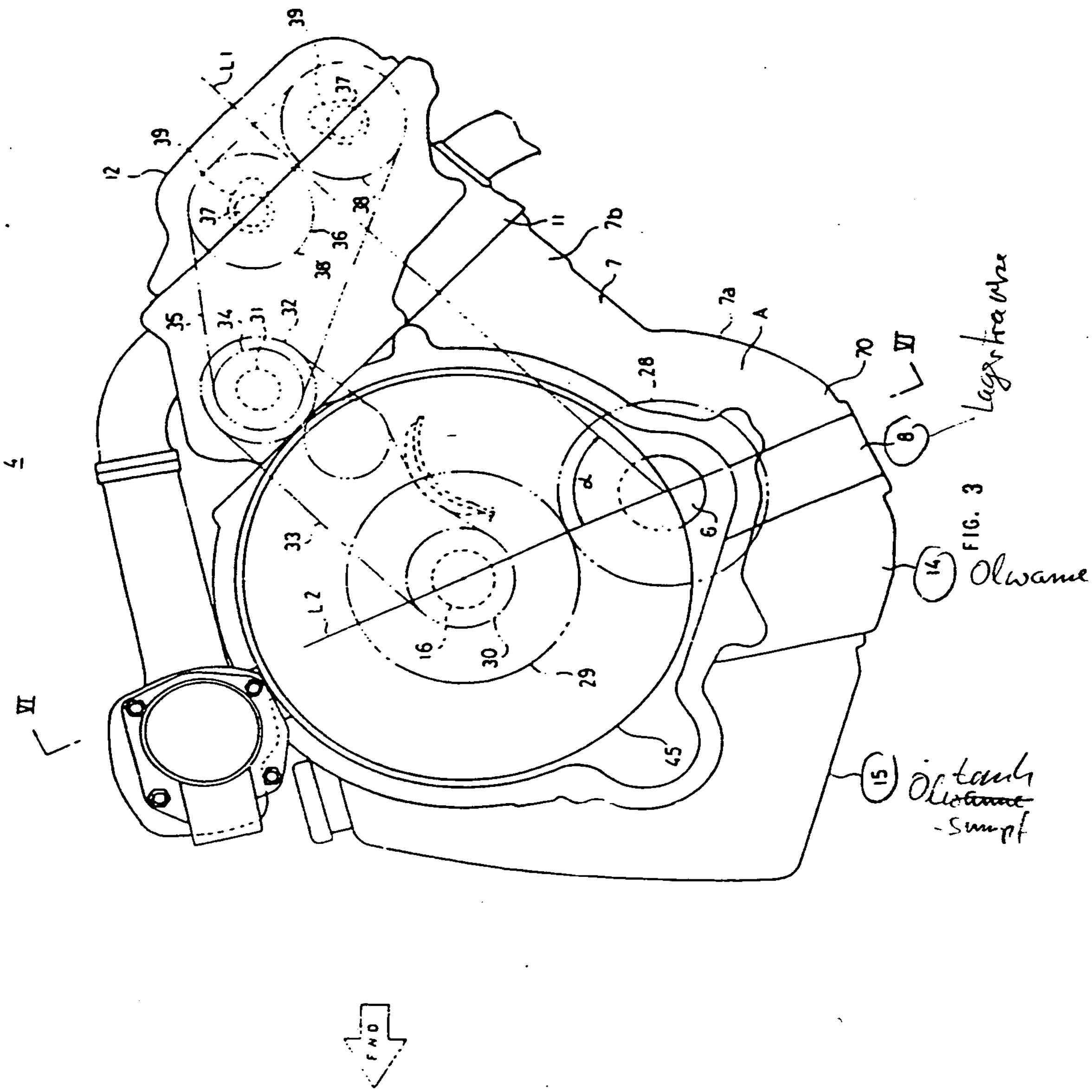


FIG. 2



4
12
39
39
L1
37
37
38
38
36
35
34
31
32
11
7b
7
7a
A
70
VI
VI
28
8
6
α
33
L2
16
30
29
45
14
15
FWD
Lagerträger
Ölwanne
Öltank
~~Ölwanne~~
-Sumpf

FIG. 3

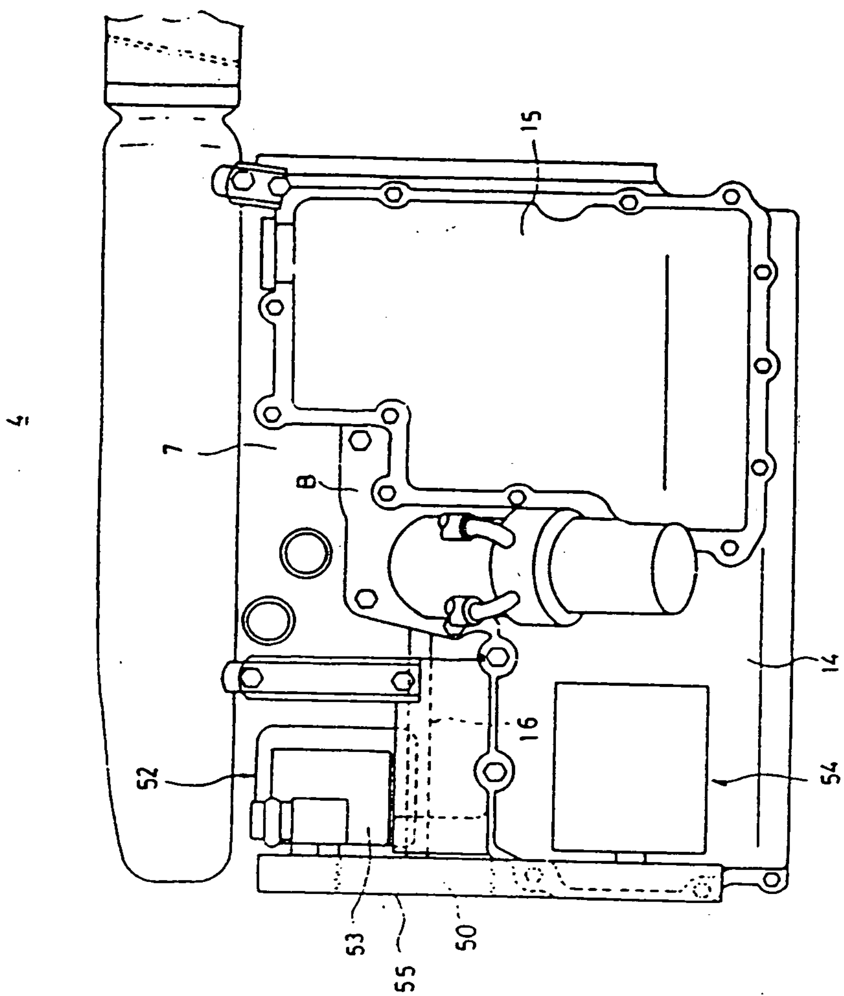


FIG. 4

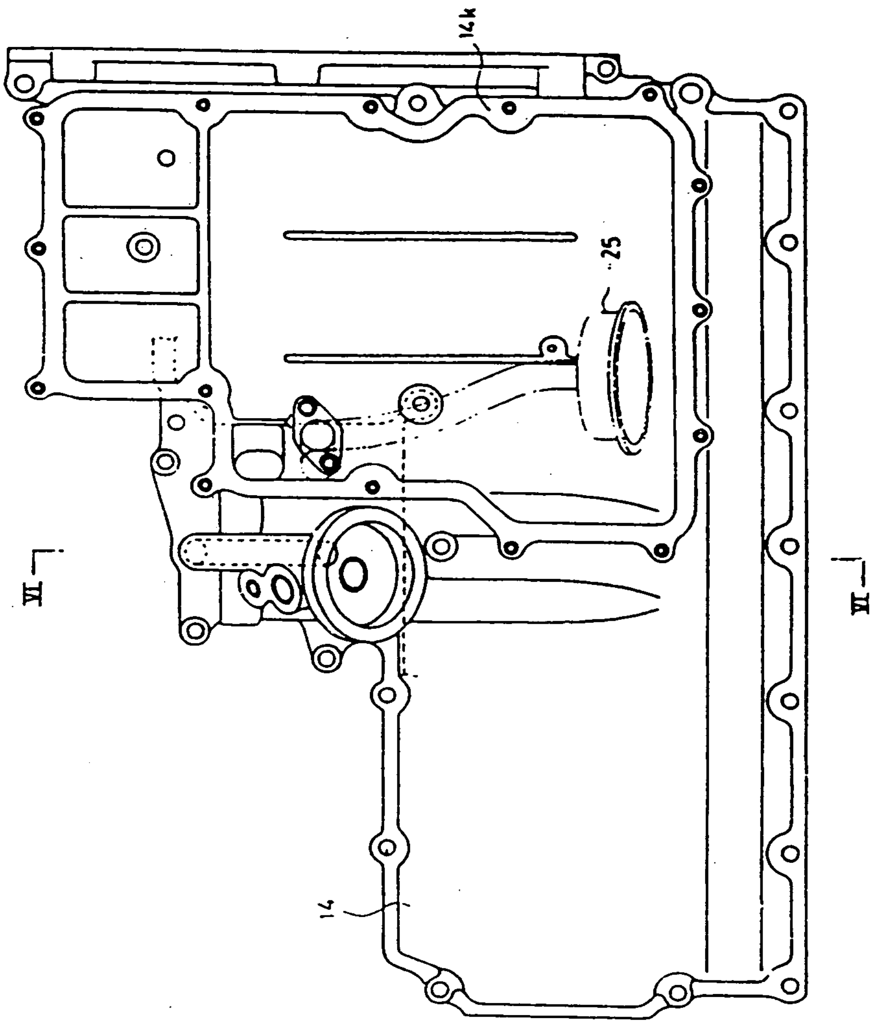


FIG. 5

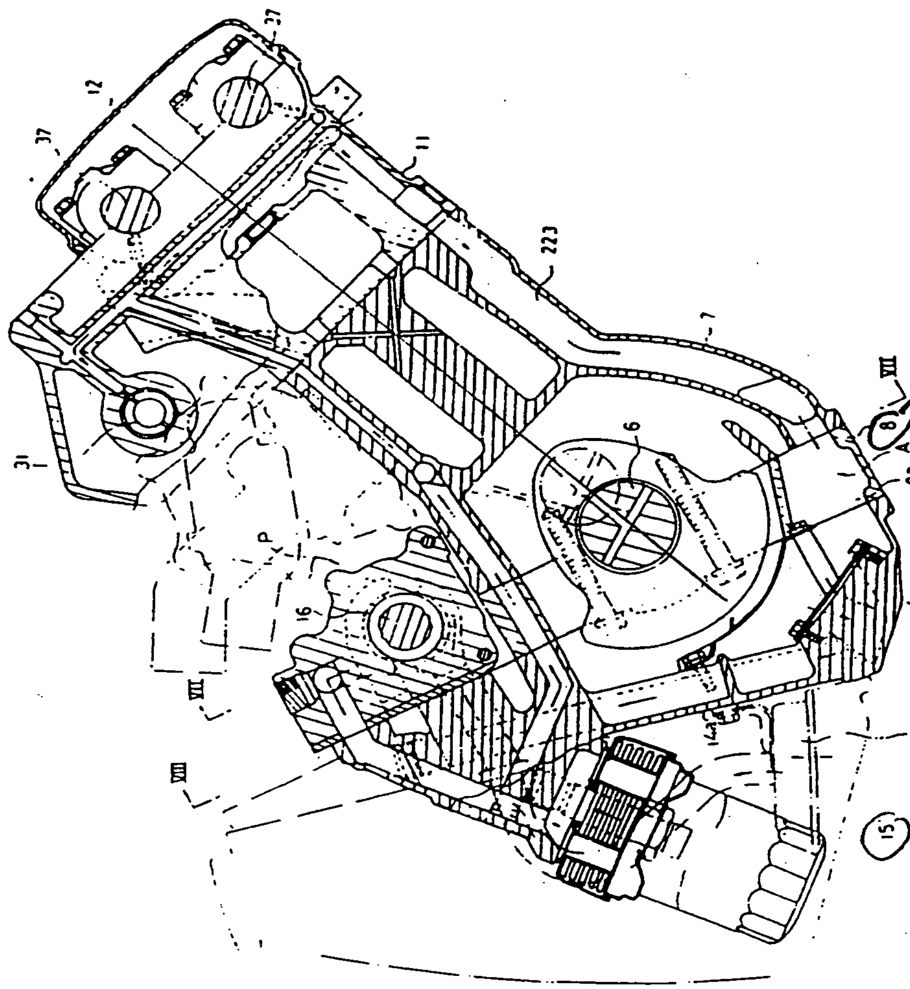


FIG. 6

Fig. 9

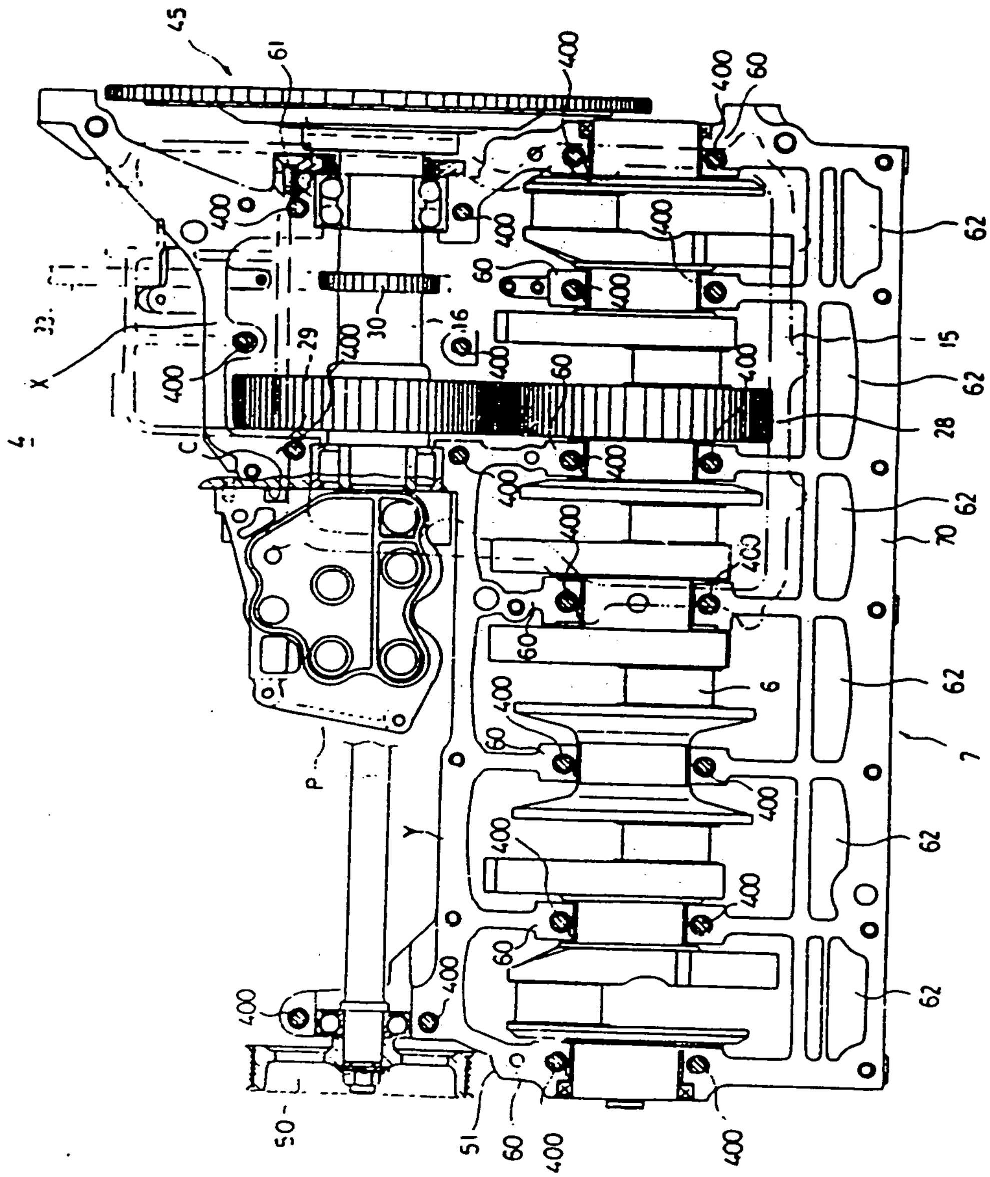


FIG.7

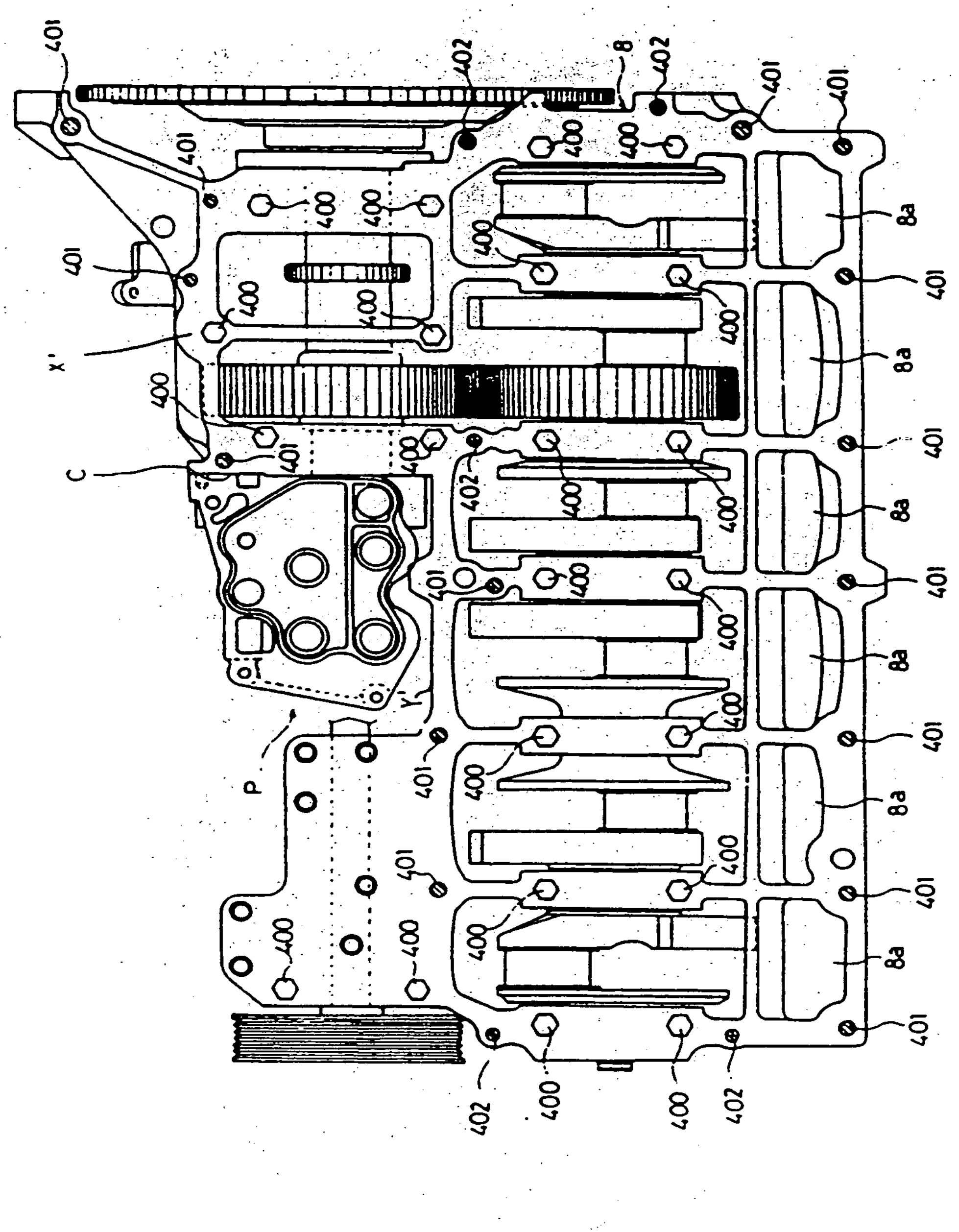

401
402
8
402
401
401
400
400
400
400
401
8a
401
400
400
400
400
X'
400
8a
401
C
401
400
402
400
400
8a
401
401
400
400
8a
Y'
P
401
400
400
8a
401
401
400
400
400
400
8a
401
402
400
400
402

FIG. 8

(wofür?)

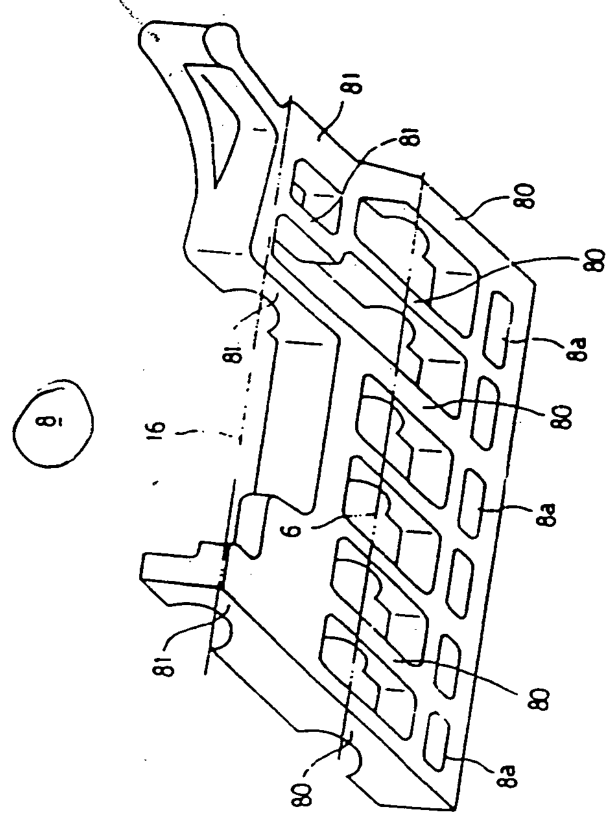


FIG. 9

Traverse für KW und Nebenwelle

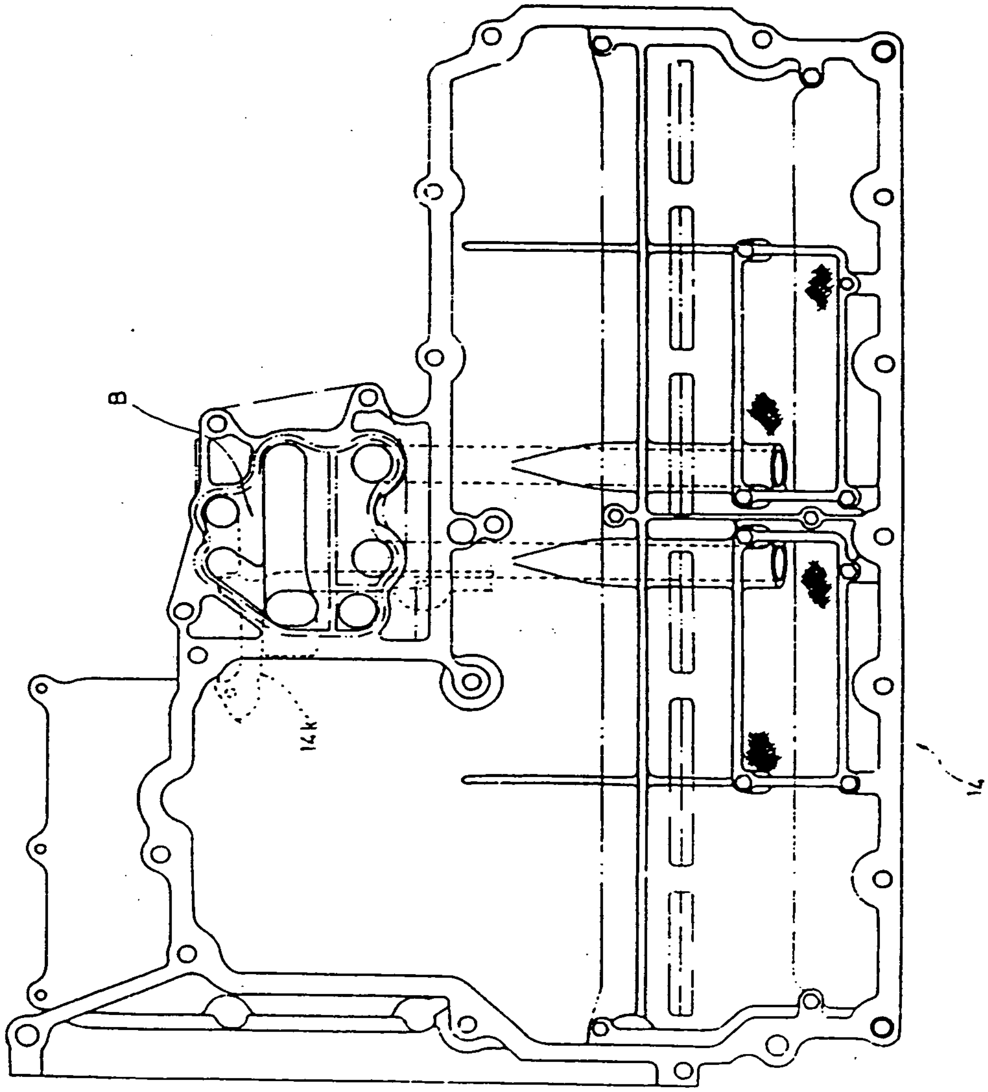


FIG. 10

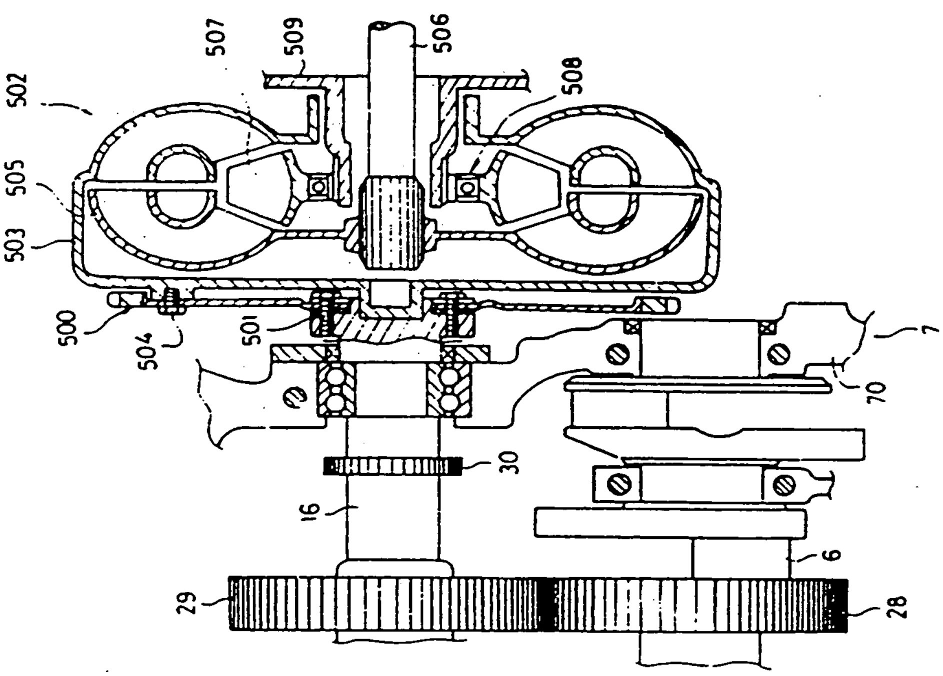
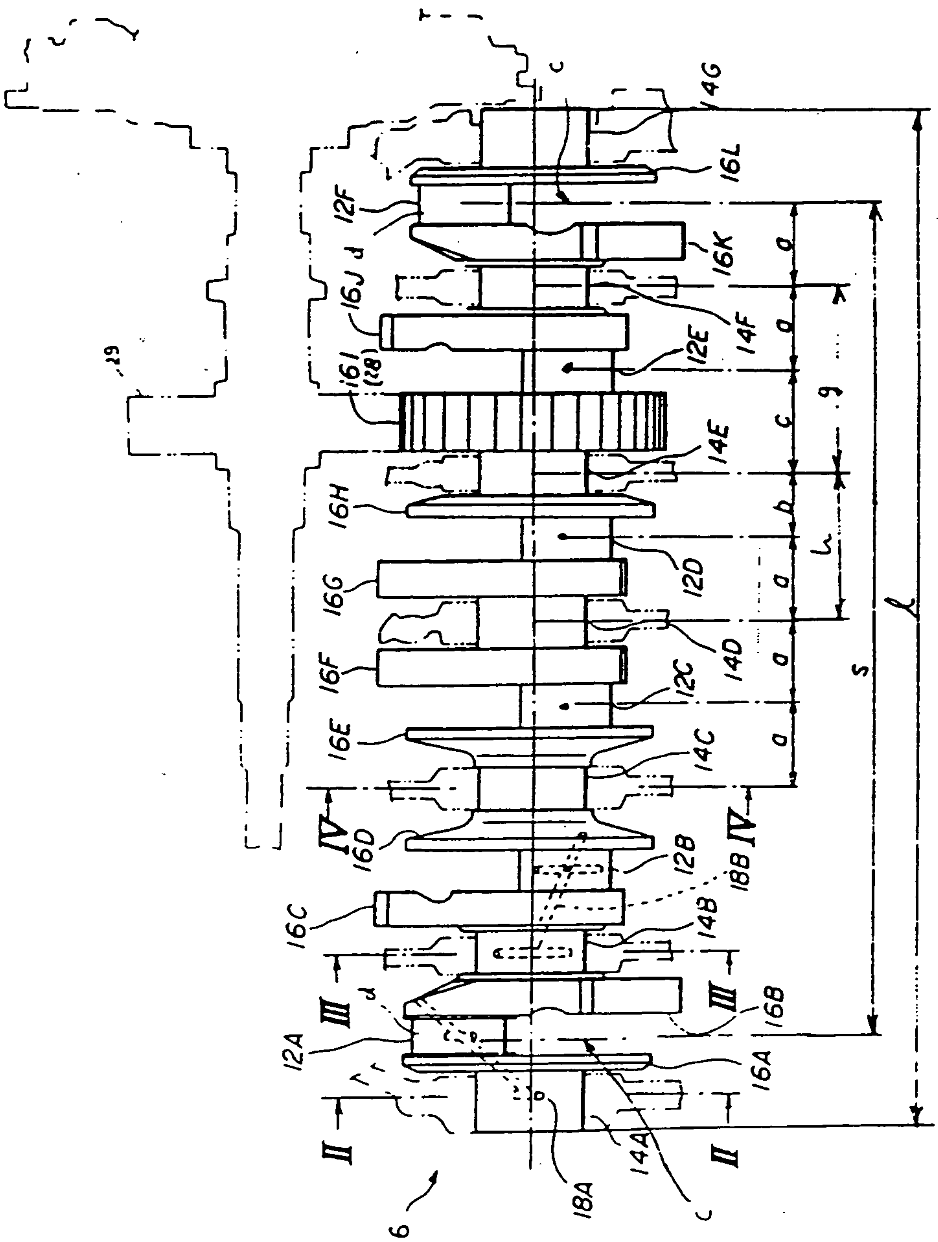


FIG. 11

Fig 12


14G
16L
12F
16K
16J
16I
(28)
29
14F
12E
14E
16H
12D
16G
16F
14D
12C
16E
14C
IV
16D
12B
18B
16C
14B
III
12A
16A
16B
II
14A
18A
6
a
b
c
g
h
s
ℓ

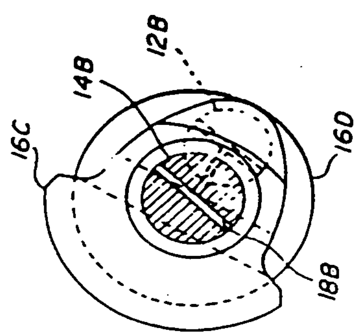


Fig. 14

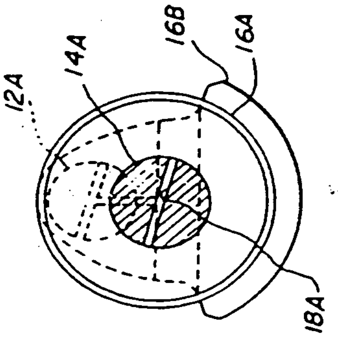


Fig. 13

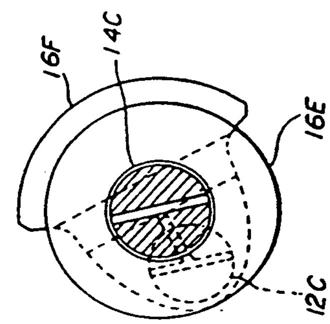


Fig. 15